INVESTIGATION OF HEAT TRANSFER AUGMENTATION THROUGH USE OF INTERNALLY FINNED TUBES

Warren Dean Snider

May 1968

INVESTIGATION OF HEAT TRANSFER AUGMENTATION THROUGH USE OF INTERNALLY FINNED TUBES

by

WARREN DEAN SNIDER

LIEUTENANT, UNITED STATES COAST GUARD

B.S., United States Coast Guard Academy

(1963)

Submitted in Partial Fulfillment of the

Requirements for the Degree of

Master of Science in Mechanical Engineering and the

Degree of Master of Science in Naval Architecture

and Marine Engineering

at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

May, 1968

INVESTIGATION OF HEAT TRANSFER AUGMENTATION THROUGH USE OF INTERNALLY FINNED TUBES

by

WARREN DEAN SNIDER

Submitted to the Department of Naval Architecture and Marine Engineering on May 17, 1968 in partial fulfillment of the requirement for the degrees of Master of Science in Mechanical Engineering and Master of Science in Naval Architecture and Marine Engineering.

ABSTRACT

Experimental data were obtained on heat transfer and pressure drop for water flowing in internally finned tubes. The test sections were steam heated copper tubes with integral internal fins ranging in size from 0.625 in. to 1.060 in. outside diameter with both straight and spiral fins. Heat-transfer coefficients and friction factors were determined for non-boiling forced convection heating.

Improvements in heat transfer for finned tubes compared to conventional tubes of the same internal diameter of up to 200 percent were found in some of the cases tested. The major portion of the improvement in heat transfer results from additional heat-transfer area added on the water side of the tube. Swirl flow was also found to contribute significantly to improved heat-transfer performance.

This investigation represents a start toward the development of a general prediction method for internally finned tubes. Further investigations are recommended to determine the effects of tightness of twist, number of fins, fin height, and fin profile on heat-transfer improvement and the value of internal fins as an augmentative technique.

Thesis Supervisor: Arthur E. Bergles
Title: Associate Professor of Mechanical Engineering

ACKNOWLEDGEMENTS

This study was supported by the F. W. French Tube Company, Newtown, Connecticut, and the finned tube specimens were supplied by that firm.

Mr. Fred Johnson of the MIT Heat Transfer Laboratory lended valuable assistance with construction details.

Mr. Richard Simonds cooperated in the construction of the apparatus and rendered valuable assistance with data reduction and other details of the project.

Miss Rita Falco typed the final manuscript.

The author's studies at MIT have been supported by the United States Coast Guard.

Grateful thanks are extended to all those concerned for their assistance.

TABLE OF CONTENTS

	PAGE
TITLE PAGE	1
ABSTRACT	2
ACKNOWLEDGEMENTS	3
TABLE OF CONTENTS	4
LIST OF FIGURES	6
NOMENCLATURE	7
I. INTRODUCTION	9
A. Heat Transfer Augmentative Techniques	9
B. Previous Investigations of Circular Tubes with Internal Fins	12
C. Purpose of Study	13
II. PROCEDURE	16
A. Scope of Research	16
B. Description of Test Loop	16
1. Water Loop	16
2. Steam Supply and Condenser	18
3. Instrumentation	20
C. Description of Test Sections	22
D. Experimental Procedure	25
E. Data Reduction Procedure	27
III. PRESENTATION AND DISCUSSION OF RESULTS	28
IV. CONCLUSIONS	37

	PAGE
V. RECOMMENDATIONS	38
REFERENCES	39
APPENDIX 1 - Details of Data Calculation Procedure	40
FIGURES	45

LIST OF FIGURES

Figure

1. Schematic of Water Test Loop
2. Schematic of Steam Condensation Apparatus
3. Friction Factors for Plain and Finned Tubes - Isothermal Conditions
4. Friction Factors for Plain and Finned Tubes - Heated Conditions
5. Correlation of Nonisothermal Friction Factors
6. Nusselt Numbers for Plain and Finned Tubes (Based on Inner Diameter)
7. Nusselt Numbers for Plain and Finned Tubes (Based on Equivalent Diameter)
8. Dimensionless Correlation of j-Factor for Internally Finned Tubes, Hilding and Coogan (4)
9. Gain in Heat Transfer of Finned Tube over Smooth Tube

NOMENCLATURE

A = area

A_c = flow cross sectional area

A_f = fin area

A_u = unfinned wall area

C_p = specific heat at constant pressure

D = diameter

D_e = equivalent or hydraulic diameter = $\frac{4A_c}{P}$

f = Fanning friction factor

G = mass velocity

h = heat transfer coefficient

K = thermal conductivity

L_h = axial heated length

L_p = axial length between pressure tops

P = wetted perimeter

p = pressure

Q = volume flow rate = (w/ρ)

q = rate of heat transfer

q/A = heat flux

r = radius

T = temperature

w = mass flow rate

η = fin effectiveness

ρ = fluid density

μ = dynamic viscosity

DIMENSIONLESS GROUPS

Nu = Nusselt number = hD/k

Pr = Prandtl number = $C_p\mu/k$

Re = Reynolds number = GD/μ

SUBSCRIPTS

b = bulk fluid condition

i = inside tube condition

in = bulk water inlet

iso = isothermal condition

o = outer wall

out = bulk water outlet

w = inner wall condition

I. INTRODUCTION

A. Heat Transfer Augmentative Techniques

Most of the effort to date in the field of heat-transfer research has been directed toward understanding the process under normal conditions. More recently, requirements for more efficient heat-transfer systems have lead to increased interest and investigation of techniques which augment or intensify heat transfer. Laboratory investigation of these techniques has reached the point where many of them may begin to be seriously considered for application to heat-exchange equipment on a commercial basis.

Techniques which have been found to improve heat transfer generally fall into one of the following categories:

1. Surface promoters
2. Displaced promoters
3. Vortex flows
4. Surface vibration
5. Fluid vibration
6. Electrostatic fields
7. Fluid additives

These techniques increase heat transfer, but usually at the expense of added pumping power, externally applied power to the system, and/or increased cost and weight.

A detailed survey and evaluation of these methods is given by Bergles and Morton (1). One of the problems is to establish generally-applicable selection criteria for the use of augmentative techniques. This appears to be a near-impossible task because of the large number of factors which enter into the decision problem. Many of these are economic factors, such as initial cost, maintenance cost, development cost, etc., and other considerations such as reliability and safety also enter the picture. Generally, those techniques requiring external power (methods 4, 5, and 6 of those listed) do not appear to be economically attractive when applied to large-scale heat-exchange equipment. Addition of small amounts of fluid additives is of benefit only in some specific applications such as saturated pool boiling with small diameter tubular heaters.

The methods requiring no external power (methods 1, 2, and 3) can be incorporated into the design of new equipment or used as a modification of existing equipment. There are difficulties connected with the installation of displaced promoters inside tubes, and they are not particularly attractive when compared to empty tubes on the basis of constant pumping power. We are left then with surface promoters and vortex flows as the two

most attractive candidates for commercial use at the present time.

Surface promoters include both the use of surface finish and finned surfaces which increase the heat-transfer area. Finned surfaces have been fairly extensively investigated for both single-phase flow and boiling heat transfer, but most of this work has been done for external fins and unfortunately is not applicable to internal fins.

Vortex flow augmentative techniques include coiled tubes, propellers, inlet vortex generators, and twisted tapes. Of these, twisted tapes appear to be the most attractive for commercial application. Detailed test results of twisted tape investigation are reported by Lopina and Bergles (2). One of the reasons for improved performance of twisted tapes is the fin effect of the tape.

It is now possible to manufacture tubes with integral internal fins in a variety of geometries. The fins can be spiraled to produce a vortex motion, and the fin effect with an integral fin is more pronounced than with a twisted tape installed within a tube due to absence of contact resistance at the tape-tube junction. There appears, then, the possibility that spiraled internal fins represent a superior augmentative technique.

One possible application of internally finned tubes is to steam condensers. Heeren (3) suggests that such tubes offer promise of substantial improvement in performance of large power plant steam condensers by reduction of the pressure differential between turbine exhaust and air-cooler section of the condenser. Internally finned tubes are suggested as one answer to the dilemma raised when inadequate space requires a compact condenser and yet efficiency must be maintained. Since these tubes have an extended surface on the water side, they are especially advantageous in the case of steam condensers or low-pressure feedwater heaters where condensing saturated steam gives a high heat-transfer coefficient on the outside of the tube as compared to the much lower coefficient on the water side.

B. Previous Investigations of Circular Tubes with Internal Fins

As stated previously, most of the research done on fins as an augmentative technique has been devoted to external fins. Circular tubes with internal fins have not been extensively investigated. Hilding and Coogan (4) presented heat-transfer and friction data for a number of internal fin configurations, including discontinuous and spiralling fins. They found continuous, straight fins to be generally quite favorable, with the

highest performance being given by those tubes with the largest inner surface area. Heeren and Wegscheider (3) reported results of efforts to develop an optimized internally finned tube for steam-condenser applications. Installation and performance of internally finned admiralty tubes replacing smooth carbon steel tubes in an existing feedwater heater are described. Lavin and Young (5) found various types of internal fins effective in increasing heat transfer in refrigerant evaporator tubes.

The studies to date on internal fins are pretty well scattered and it appears that no attempts have been made to develop a general prediction method. With an infinite number of possible fin configurations, testing them all appears to be out of the question. As a practical matter, development of a prediction method would therefore seem to be a worthwhile goal.

C. Purpose of Study

In order to better evaluate the worth of integral internal fins in tubes as an augmentative heat transfer technique, the goals of this study were defined to be:

1. Construct an apparatus suitable for measurement of heat-transfer coefficients for flow inside tubes. The nature of the tube specimens required steam heating in lieu of electric heating.

2. Obtain heat-transfer and pressure-drop data for various tube geometries and flow conditions and

compare these to conventional empty tube data.

3. Evaluate the data and attempt to develop an analytical approach to predicting performance of other geometries.

II. PROCEDURE

A. Scope of the Research

A comprehensive investigation was made of the forced-convection, nonboiling heat-transfer coefficient and pressure drop for the heating of water flowing through internally finned tubes of three different geometries and one smooth tube. The range of variables investigated and the design of the steam-heated test sections and condensing apparatus were dictated largely by the capabilities of the MIT Heat Transfer Laboratory low pressure water test loop and the available steam supply. Basic system variables were:

Inlet temperature: $50^{\circ}F$ - $120^{\circ}F$

Reynolds Number at test section inlet: 360 - 31,200

Maximum heat flux attained: 101,000 $Btu/hr\text{-}ft^2$

B. Description of Test Loop

1. Water Loop

The water test loop used in this study was located in the MIT Heat Transfer Laboratory. It was designed and constructed in 1961 and its construction and operation are fully described in references (2) and (6). A schematic diagram is shown in Fig. 1. It is a closed loop, low pressure system which circulates distilled water through the test section. A by-pass line permits control of flow rate through the test section by allowing a portion of the pump discharge to recirculate,

bypassing the test section. The loop is provided with filters and an ion-exchange demineralizer to maintain the water free from dissolved minerals and other particulate matter, a degassing tank which also serves as a surge tank, an accumulator to reduce pressure fluctuations, rotameter and metering valve to measure and control flow through test section, preheaters to adjust inlet temperature of water, and a heat exchanger which cools loop water by use of fresh water from the city water supply.

Rate of fluid flow through the test section and test section inlet pressure are controlled by adjusting the metering valve at the test section inlet, the valve in the bypass line, and the test-section exit valve.

Both forced-convection and two-phase-flow experiments have been carried out on this loop in the past, but until now all test sections have been electrically heated by passing a current through the tube walls. In this investigation, all of the tests were made for forced-convection conditions and most for inlet temperatures on the order of $60^{\circ}F$. These conditions are fairly representative of those that might exist in steam condenser applications. No two-phase-flow tests were performed. For the tests made, electrical heating of the test sections was not practical, and this necessitated construction of a suitable piece of test apparatus.

2. Steam Supply and Condenser

Because of the non-uniform nature of the walls of the tubes to be tested and the possible variation of heat flux at the location of thermocouples embedded in the tube wall if the tube itself is used as a heater, dropwise condensation of steam was selected as the preferable method of heating the test section. Dropwise condensation offers advantages over filmwise condensation; condensation is more nearly uniform all around the perimeter of the tube with dropwise condensation, and the dropwise condensation heat-transfer coefficient is on the order of ten times as large as either the filmwise condensation coefficient or the water-side heat-transfer coefficient.

Since existing test facilities did not permit steam heating of the test sections, design and construction of a suitable apparatus to be used in conjunction with the existing water loop was required. This constituted the first portion of the required work, and the result now serves as a piece of auxiliary equipment to the basic loop which extends its utility and paves the way for continued future investigations where steam-heated test sections are required.

The apparatus developed is essentially a one-tube shell-and-tube condenser with a shell diameter of four

inches, a length of three feet, and capable of accepting tubes of up to 1.110 in. outside diameter. A schematic of the steam condensation rig is shown in Fig. 2. The description given by Sachs (7) for a similar apparatus was used as a guide in early stages of design. Steam is provided from the building heating system steam supply at 25-40 psig, depending upon the season and attendant demands upon the system. It is reduced to the desired pressure by a Spence Type EW pressure regulating valve and passes into the condenser via the steam inlet pipe. The steam inlet pipe is a 1/4-inch (nominal) copper tube with 16 holes (#50 drill) in its upper surface. The steam inlet pipe runs axially the length of the condenser near the top of the condenser shell. The total area of the holes equals 90 percent of the internal cross-sectional area of the steam supply tube to insure even distribution of steam inside the shell. Condensate passes out of the shell through a drain fitted with a thermostatic trap.

A 2 1/2-inch-diameter sight port was installed in the condenser shell at its midlength to permit observation of condensation on the test section outer wall to determine if it was dropwise or not. To reduce condensation on the inside of the sight port and permit clearer observation, a portable heat gun was used to blow hot air on the glass port.

The test section and condenser shell surrounding it, along with associated steam and drain piping, valves, gages, etc., are mounted on a portable test stand on casters. This stand can be wheeled up to the low pressure water test loop and connected to it by flexible hoses. A 20-foot steam hose supplies steam from the building steam system. This portable arrangement reduces interference among experiments concurrently in progress or being set up on the loop, and permits work on the condenser stand in a more convenient and less congested portion of the lab than that immediately adjacent to the water loop.

Following completion of construction, the steam side of the condenser (including sight port) and associated steam and drain piping were hydrostatically tested to a pressure of 60 psig for a period of 10 minutes. This is approximately 1.5 times the maximum steam supply pressure, and 20 times the normal operating pressure. The test was intended to verify the integrity of the condenser shell and assure safety for the operator.

3. Instrumentation

The following system parameters were measured and recorded:

mass flow rate

temperatures

test-section inlet

test-section outlet

tube wall temperatures (5)

pressure at test-section inlet

pressure drop across test section

Rotameters (Fischer-Porter Flowrators) with overlapping ranges from 15 to 4,000 lbm/hr were used to measure the flow rate.

Fluid temperatures were measured by 30-gage copper-constantan thermocouples connected to a common icebath, and the millivolt outputs were read on a Minneapolis-Honeywell, Brown recorder. Test-section inlet temperature was measured by a thermocouple inserted into the flow through a Conax fitting in a pipe tee attached to the inlet end of the test section. The outlet thermocouple was inserted near the exit of a mixing chamber attached to the downstream end of the test section. Three baffles within this chamber served to mix the flow so the exit temperature measured was in fact a mixed mean temperature. Tests of this chamber arrangement indicated that the temperature at the exit was uniform across the flow down to the lowest flow rates tested. Tube wall temperatures were measured by five thermocouples embedded in the tube wall and lead out through a shell penetration in the inlet header of the condenser.

Fluid pressures were measured on Bourdon-tube gages. Two Helicoid 8 1/2 in. gages were used for test section

inlet pressure and condenser shell steam pressure. Pressure drop across the test section was measured with a 60 in. Meriam U-tube manometer filled with Meriam Fluid No. 3 (specific gravity of 2.95) and a 30° inclined manometer filled with the same fluid and covering a two-inch range.

C. Description of the Test Sections

Test section 1 was a locally-procured commercially-drawn copper tube. The remainder of the tubes were specimens provided by the F.W. French Tube Co., Division Valley Metallurgical Processing Co., Inc., Newtown, Connecticut, and are of the type referred to by them as FORGE-FIN. These tubes have integral internal fins and can be produced by the manufacturer in a wide variety of geometries, including straight or spiral fins of various heights and profiles, and duplex or triplex tube-within-a-tube arrangements. All of the tubes tested were of copper. The fin configurations tested are described in Table 1 and included quite a wide range of geometries.

Instrumentation of the test sections included installation of bushings where the tube passed through the condenser shell, thermocouples to measure tube wall temperatures, and pressure taps at inlet and outlet ends of the heated length. This was followed by a thorough cleaning of the test section and application of the promoter used to insure dropwise condensation.

TABLE 1

DETAILS OF TUBES

Tube	1	3	4	5
Description*	Plain	Spiral Fin	Straight Fin	Spiral Fin
Material	Copper	Copper	Copper	Copper
Total length of tube, ft	5.80	5.37	5.96	5.96
Heated length, ft	3.04	3.00	3.00	3.00
L/D_i	67.6	36.5	65.0	62.0
Unheated entrance length, ft	2.28	1.86	2.42	2.42
L/D_i	50.8	22.6	52.5	50.0
Inner diameter, ft	0.0449	0.0822	0.0462	0.0484
Outer diameter, ft	0.0520	0.0884	0.0520	0.0520
Tube wall thickness, ft	3.68×10^{-3}	3.08×10^{-3}	3.00×10^{-3}	1.75×10^{-3}
External surface area, ft^2	0.490	0.834	0.490	0.490
Cross-sectional area, ft^2 (flow area)	1.575×10^{-3}	4.921×10^{-3}	1.664×10^{-3}	1.746×10^{-3}
Wetted perimeter, ft	0.141	0.434	0.232	0.246
Equivalent diameter, ft	0.0449	0.0458	0.0287	0.0284
Fin efficiency,	0	0.955	0.932	0.970
Internal surface area, ft^2	0.424	1.292	0.698	0.737
Unfinned area A_u	0.424	0.525	0.317	0.232
Fin area A_f	0	0.767	0.381	0.505

*Tube 3 16 spiral fins, 0.080 in. high, 360° twist in 6.25 in. length, trapezoidal profile

Tube 4 10 straight fins, 0.060 in. high, trapezoidal profile

Tube 5 30 spiral fins, 0.030 in. high, 360° twist in 7.0 in., triangular profile

In order to facilitate the testing of tubes of various sizes, the penetrations in the blank flanges which constituted the headers or end bells at either end of the condenser shell were made large enough to accept tubes up to 1.110 in. and fitted with Goddard fittings which seal by compression of a neoprene "O" ring against the tube. The tube outside diameters were bushed up to this size by brass sleeves in the case of the smaller tubes and Scotch No. 67 electrical tape (epoxy film reinforced with glass cloth) in the case of the largest tube. This arrangement, coupled with the location of the penetration required for the thermocouple wires, facilitated testing tubes of all sizes less than 1.110 in. and made the task of changing tubes less troublesome.

Five thermocouples were used to measure tube wall temperature. These thermocouples were soldered at equal intervals along the tube heated length in slight depressions or pits of approximately 0.020 in. depth drilled in the tube wall. Since the thermocouple beads were approximately 0.030 in. to 0.035 in. diameter, they were not truly embedded; the pits served essentially as an aid to soldering the thermocouple onto the tubes. After soldering, the thermocouple attachment points were covered with Hysol epoxy to protect them from damage

during further preparation of the test section. The thermocouple wires were led through a Conax fitting with a crushable lava sealant in the inlet header of the condenser shell, and from there to the recorder via a switching box.

Pressure taps were installed on the test section just outside the inlet and outlet headers of the condenser shell. These were made from 1/8-to-1/4-in. N.P.T. reducers soldered to the tube with a 0.0210 in. hole drilled through the tube wall and deburred on the inside.

Before installation, test sections were cleaned and degreased, and then a two percent (by weight) solution of di-n-octadecyl disulfide in carbon tetrachloride was applied from a wash bottle. This, when dry, left a rather nonuniform, chalky residue on the tube which went away shortly after condensation began within the condenser shell. Presumably, a monolayer thickness remained to perform the promoter function, since the condensation observed was in fact dropwise.

D. Experimental Procedure

After the test section was installed in the condenser, connections were made to loop cooling water and steam supply, the thermocouples were connected to the recorder selector switch, and pressure gages and manometers connected up. The system was filled from the

degassing tank and supply tank and air bled from high points of the system.

For isothermal pressure drop runs, the main loop throttle valve was adjusted to give the desired pressure and the metering valve at the test-section inlet used to control flow rate through the test-section. The heat exchanger and preheaters were used as necessary to control inlet temperature of the water. System pressure, pressure drop across the test section, flow rate, and fluid temperature were recorded.

For heating runs, the bypass valve was adjusted to attain the desired inlet pressure, flow rate set at the desired quantity, and steam supply cut in. The regulating valve was adjusted to maintain steam pressure in the condenser shell at about 2 1/2 psig which gave a shell temperature of about 220°F. At very high flow rates with low inlet temperatures, the condenser capacity exceeds the steam supply from the regulating valve, and steam pressures are consequently lower. An air vent served to carry off excess steam and any air and noncondensible gases; the vent was normally left open enough to provide a good healthy wisp of steam to insure that noncondensible gases were swept out of the condenser. Early in the project some tests were run to insure that flow through the condenser was adequate to prevent buildup of air within the condenser shell and

consequent reduction in heat transfer. These tests failed to give any sign of problems with air accumulation. After the system had reached equilibrium (exit and wall temperatures reached steady values), flow rate, fluid and tube wall temperatures, inlet pressure, and pressure drop were recorded. Steam shell temperature and pressure were also recorded; although they were not needed in reduction of the data, when coupled with the condensate flow rate they do serve as a check on condenser operation.

E. Data Reduction Procedure

A FORTRAN IV computer program was used in conjunction with the Mechanical Engineering Department's IBM 1130 computer to reduce data for tube number one, and to provide partial data reduction for the finned tubes. The remainder of the data reduction for the finned tubes was carried out by hand calculation. Details of the calculation procedure are presented in Appendix 1.

III. PRESENTATION AND DISCUSSION OF RESULTS

The experimental forced-convection heat-transfer and pressure-drop results are presented in Figs. 3-7. Pressure-drop data are presented with friction factor plotted against Reynolds number, where both are defined in terms of the hydraulic or equivalent diameter of the tube - four times the cross-sectional area divided by the wetted perimeter. All properties are evaluated at the fluid bulk temperature.

Data for isothermal friction factors generally follow the pattern one would expect in the turbulent region. Friction factors for the plain tube are higher than the commonly-used correlation for smooth tubes (8), but those for the others are still higher, although all lie surprisingly close together. Several factors may be working together to cause the observed increase in friction factor: tube roughness, swirl flow, and the possible inadequacy of the equivalent diameter concept are all involved here. Roughness measurements of tube walls and fin surfaces were planned to give some idea of the roughness contribution, but these were delayed due to repairs on the Talysurf roughness measuring apparatus and could not be completed in sufficient time to be included here.

Data in the transition and laminar regions exhibit considerably more scatter than that evident at higher flow rates. Part of this may be due to inaccuracies associated with measuring small pressure drops; data in these regions correspond to differences in height of oil columns on the order of only 0.10 in.

Friction factors for heated flow are lower than those for isothermal flow, as one would expect, due to lower viscosity in the vicinity of the heated wall. Figure 5 presents a correlation of the nonisothermal flow friction factors of the form $f = f_{iso}(\frac{\mu_w}{\mu_b})^{\alpha}$ where μ_w is evaluated at the wall temperature and μ_b is evaluated at the bulk fluid temperature. Most of the data are correlated for values of α between 0.14 and 0.35. Shortcomings in the ability of the manometer system to accurately measure small pressure drops produced some rather erratic results here as in Figs. 3 and 4.

Heat-transfer results are presented in Fig. 6 on the coordinates of $Nu_i/Pr^{0.4}$ versus Re_i. These coordinates were based on the inside diameter in order to show the improvement obtainable with a finned tube over a comparable empty tube. In this case, the heat-transfer coefficient is defined by the equation

$$h = \frac{q/A_i}{T_w - T_b} \qquad (1)$$

where $A_i = \pi D_i L_h$ and T_w and T_b are computed by the averaging technique outlined in Appendix 1.

In Fig. 7 heat-transfer results are presented with Nu and Re defined in terms of equivalent diameter of the tube. Since the equivalent diameter is roughly half the inner diameter for each of the tubes tested, both Nu and Re are reduced by approximately half from Fig. 6 to Fig. 7.

Here the heat-transfer coefficient is defined by

$$h = \frac{q/A_{eff}}{T_w - T_b} \tag{2}$$

where $A_{eff} = A_u + \eta A_f$, and η is a performance factor or fin efficiency and defined as the heat transfer rate from the fin divided by the heat transfer rate if the fin were uniformly at the base or wall temperature.

The high heat transfer for the smooth tube at Reynolds numbers below 2700 may be due to increased secondary flow at low flow velocities. Axial flow velocities in this region are less than about 30 ft/min. A somewhat similar phenomenon was reported by Newell (9) in a Reynolds number range of 2000-9000 for flow in annular channels, where an increase in heat transfer was attributed to strong secondary flow patterns and the possibility that such flows induced a transition from laminar to turbulent flow. In a subsequent analytical study, Newell (10)

showed that heat-transfer coefficients for fully-developed laminar flow can be well over 100 percent above the conventional solutions when buoyant effects are considered.

Figures 5 and 6 indicate that finned tubes offer improvement in heat transfer, both when the comparison is made to an unfinned tube of the same inner diameter and when the finned tube is compared to one of the same equivalent diameter.

The improvement in heat transfer may generally be attributed to the following factors:

1. Roughness of inner surface finish - the Forge-Fin process generally leaves the surface with some marks
2. Swirl flow induced by spiral fins
3. Secondary flow due to natural convection
4. Complex geometry effects which cannot be correlated in terms of the equivalent diameter

Let us turn our attention to Fig. 7. Visually, all of the finned tubes appear to be of roughly comparable surface finish. Talysurf measurements would, of course, give more precision to this evaluation, but gross differences are clearly not present. In view of this, it would appear doubtful that roughness alone could account for the difference in performance among the tubes. So, although surface roughness may contribute to improvement,

it is probably not too significant in this case. Likewise, whatever natural convection or secondary flows occur, they are limited to comparatively low flow rates. That leaves swirl flow and complex geometry as the main factors to be considered.

It appears that swirl flow may be at least part of the answer. The twist of the fins is tabulated below:

Tube	Twist	No. of $D_e/360^o$
1	no fin	
3	360^o/6.25 in.	11.6
4	none	
5	360^o/7.0 in.	20.6

Thus tube 3 has the tightest twist (when compared in this fashion at least), and the "degree of twist" corresponds roughly to the degree of improvement. It is significant to note that while the difference between the performance of 3 and 5 is not marked, they both are significantly better than the tube with straight fins.

The remaining factor in the explanation of improvement in heat transfer is the complex geometry of the tube. Is the definition of equivalent diameter adequate for these tubes? What effect does fin size have on improvement? These are, unfortunately, things upon which tests of three tubes of differing size and fin geometries are hardly an adequate data base upon which to offer firm

conclusions. The question of adequacy of equivalent diameter is somewhat bothersome. While it has been shown adequate for rectangular channels, and a similar concept is used for annular flow channels, these certainly do not approach the rather tortured geometry of these tubes. However, in the absence of anything better or occurrence of obvious difficulties, perhaps the only sensible course is to assume it is adequate and proceed on that assumption. The sorting out of the effect of various parameters of the fin geometry (number of fins, fin height, fin shape, twist) will require further tests in which just one parameter is varied at a time.

It is interesting to compare heat-transfer results shown in Figs. 6 and 7 to results of previous investigations. Hilding and Coogan (4) reported heat-transfer results for a number of straight and discontinuous fin configurations with air. Results for fin geometries similar to those investigated here are repeated in Fig. 8 plotted on coordinates of j versus Re, where the heat transfer j-factor (or Colburn Number) is defined by

$$j = St \cdot Pr^{2/3} = \frac{h_f}{GC_p}\left(\frac{C_p \mu_m}{k_m}\right)^{2/3}$$

Here m refers to mean fluid properties and h_f is the average air-side surface coefficient for the exposed

heating surface including fin surface. Reynolds number is based on equivalent diameter. For all the fin designs tested, including those shown, Hilding and Coogan found a decrease in the j factor as compared to the unfinned tube at Re = 5000. At lower Reynolds numbers, the j factor is greater in some cases and less in others. The H7-3 configuration (four full fins) gave the best results of the continuous fins tested and this produced a maximum improvement of about 20 percent at Re = 3500. This decreased to zero at Re = 7000, and the heat transfer is not as good for the finned tube above that point.

Heeren and Wegscheider (3) found improvements in heat transfer coefficients in tubes with internal fins of about 40 to 100 percent over conventional smooth tubes. Gain in heat transfer coefficient of finned tube over smooth tube is shown as a function of water velocity in Fig. 9 for the 23-20 type tube (approximately 7/8 in. outside diameter) of Heeren and Wegscheider. Also plotted is a similar gain of the 1.060 in. Forge-Fin tube (data from Fig. 6). Since the Forge-Fin tube has fins with a relatively tight spiral, while the 23-20 type tube has straight fins, the results of the comparison are not unreasonable.

This investigation represents a start toward developing a general prediction method for internally finned tubes. Results of tests for three tubes is insufficient

data upon which to base such a prediction. It does appear that swirl flow contributes significantly to improvement in heat transfer. Comparison to the improvement reported by Heeren and Wegscheider for a straight-finned tube seems to bear this out. The effects of fin geometry are not yet clear. Additional tests will be required to sort out effects of various fin geometry parameters. The contribution of surface roughness to improvement in heat transfer does not appear to be significant in the tubes tested.

Internally finned tubes do offer significant improvements in heat transfer, particularly when compared to unfinned tubes of the same internal diameter. The increase in some cases may be as great as 200 percent, due to the additional area and the effects discussed.

The improvements appear to be much more significant than those previously reported for air, and of comparable magnitude to those reported for water.

We have chosen to ignore here the increase in pumping power which results from the higher friction factors of internally finned tubes, in order to concentrate on improvements in heat transfer. The decision of what, if any, augmentative technique to employ must take the question of increased pump power into account if it is to be realistic. A further discussion of this point and a method for comparison are given in references (1) and (2).

Other factors in the evaluation of internal fins as an augmentative for use in tubes which have been ignored here include all those that fall into the "economic" category. A major component of these is the manufacturing cost of the tubes. The relative ease or complexity of the tube fabrication process will figure strongly in the over-all optimization. In this respect, integral internal fins offer obvious advantages over built-up tubes such as those described in (4).

Finally, as a useful by-product of the work required for this investigation, an apparatus suitable for gathering heat-transfer data on tubes where dropwise condensation of steam is the preferred method for heating the tube has been added to the MIT Heat Transfer Laboratory's water test loop, and the necessary methods and procedures for carrying out such tests have been developed and tested.

IV. CONCLUSIONS

The conclusions of this investigation on heat transfer augmentation through use of internally finned tubes can be summarized as follows:

1. Integral internal fins offer potential for improvement in heat transfer over conventional tubes of the same internal diameter. Improvements of up to 200 percent were found to be possible in some cases.

2. Improvement in heat transfer is a result of adding additional area on the water side of the tube. A significant contribution also results from swirl flow produced by spiral fins.

V. RECOMMENDATIONS

The recommendations arising out of this investigation are:

1. Additional systematic investigations of internally finned tubes should be carried out to determine the effects of various geometry parameters on heat transfer improvement. Parameters of interest include:

    a. Degree of twist

    b. Number of fins

    c. Fin height

    d. Fin profile or shape

2. Explore possibilities of better, more accurate methods of measuring pressure drop across the test section, particularly with large tubes at low flow rates.

REFERENCES

1. A.E. Bergles and H.L. Morton, "Survey and Evaluation of Techniques to Augment Convective Heat Transfer," MIT Engineering Projects Laboratory Report 5382-34 (1965).

2. R.F. Lopina and A.E. Bergles, "Heat Transfer and Pressure Drop in Tape Generated Swirl Flow," MIT Engineering Projects Laboratory Report 70281-47 (1967).

3. H. Heeren and J.J. Wegscheider, "Internally Finned Tubes, A Design Tool to Improve Condenser Performance," ASME Paper No. 67-WA/CT-2 (1967).

4. W.E. Hilding and C.E. Coogan, Jr., "Heat Transfer and Pressure Loss Measurements in Internally Finned Tubes," Symposium on Air-Cooled Heat Exchangers, ASME, 57-85 (1967).

5. J.G. Lavin and E.H. Young, "Heat Transfer to Evaporating Refrigerants in Two-Phase Flow," AIChE Preprint 21e for National Meeting (1964).

6. A.E. Bergles and W.M. Rohsenow, "Forced-Convection Surface-Boiling Heat Transfer and Pressure Drop in Tubes of Small Diameter," MIT Department of Mechanical Engineering, Report No. 8767-21, May 25, 1962.

7. P. Sachs, "Modified Correlation for the Rate of Heat Transfer to Water Flowing in a Tube," Journal Mechanical Engineering Science, 4, 1, 78-84 (1962).

8. Pipe Friction Manual, Hydraulic Institute, 61 (1954).

9. P.H. Newell, Jr., "The Effect of Untrasonic Vibration on Forced Convection Heat Transfer," MIT Mechanical Engineering Thesis, June 1964.

10. P.H. Newell, Jr., "Laminar Flow in Horizontal Tubes," MIT PhD Thesis, June 1966.

11. N. Todreas and W. Rohsenow, "Effect of Nonuniform Axial Heat Flux Distribution," MIT Department of Mechanical Engineering, Report No. 9843-37 (1965).

Appendix 1

DETAILS OF DATA CALCULATION PROCEDURE

The experimental data for heated runs on tube 1, the smooth tube, were reduced by a computer program on the IBM 1130. This program was also used to provide partial data reduction for heated runs on finned tubes. Remainder of finned tube data reduction and reduction of all isothermal data was carried out by hand.

The equations and computational methods used by this program are outlined below:

Input Information

The test section input information was as follows:

Tube number

Heated length, ft

Inner wall radius, ft

Radius from tube centerline to thermocouple position, ft

Difference between heated length and distance between pressure taps

Run input information consisted of:

Run number

T_{in}

T_{out}

$T_{wall(1-5)}$

Flow rate, w, lb/hr

pressure drop, in. of oil (S.G. = 2.95)

inlet pressure

Head Loss and Friction Factor

Head loss (in feet of water) was calculated from

$$\text{head loss (ft)} = \frac{\Delta p}{\rho g} = \frac{\text{CONST x } \Delta h}{62.0} \qquad (1)$$

where Δh = difference in monometer fluid levels

Friction factors were calculated from

$$f = \frac{\Delta p}{4(^{L}p/D_i)(\rho V^2/2)} \qquad (2)$$

Tube Wall Temperature Drop

Overall rate of heat transfer was found from

$$q = wC_p(T_{out}-T_{in}) \qquad (3)$$

Temperature at inner wall was found from the Fourier rate equation

$$T_1 = T_o - \frac{q}{2\pi k L_h} \times \ln \frac{r_o}{r_w} \qquad (4)$$

It was assumed that the overall heat transfer coefficient is constant along the tube and that the wall temperature is essentially constant along the length of the tube. This latter assumption agreed with experimental observations at all but the very highest of flow rates. For these high flow rates, the analysis given by Sachs (7) taking variation of wall temperature into account was investigated. The average of the five heat transfer coefficients produced by this method was so

close to the result obtained by averaging the five temperatures indicated by the wall thermocouples that it was decided to use the simplier method with the implied assumption that the wall temperature was constant and equal to the average of the values indicated by the thermocouples.

Heat Transfer Coefficient and Bulk Water Temperature

Both the heat transfer coefficient and bulk water temperature were computed from

$$\int_{T_{in}}^{T_{out}} wC_p dT_b = \int_{L_o}^{L_i} h_w(T_w - T_b)(\pi D)d\ell \qquad (5)$$

which integrated between inlet and outlet gives the following equation for h_w

$$h_w = -wC_p \, \ell n \left[\frac{(T_w - T_{out})}{(T_w - T_{in})} \right] / L_h(2\pi r_w) \qquad (6)$$

To find bulk water temperature, Equation (5) is solved for T_x, bulk water temperature at any point along the tube and then integrated over all x and divided by L_h to find an average T_b. The result is

$$T_b = T_w + \left[\left(\frac{T_w - T_{in}}{nY} \right) \frac{1}{Y} - 1.0 \right] \qquad (7)$$

where $Y = \dfrac{T_w - T_{in}}{T_{in} - T_{out}}$

<u>Evaluation of Fluid Properties and Dimensionless Groups</u>

The evaluation of the fluid properties as a function of temperature was accomplished by the computer program. Fluid viscosities were calculated by a power series accurate to temperatures up to 600°F used in (11). Thermal conductivities of water were calculated using a curve fit of tabulated values over the range 50°F-200°F. Thermal conductivity of copper was assumed constant equal to 219 Btu/(hr-ftF).

Dimensionless groups were evaluated as follows:

$$Nu = \frac{hD_i}{k}$$

$$Re = \frac{GD_i}{\mu}$$

$$Pr = \frac{C_p\mu}{k}$$

<u>Reduction of Finned Tube Heating Data and Isothermal Pressure Drop</u>

Heat flux, Prandtl number, and third properties computed by the program were used to compute results for finned tubes

$$\frac{Nu}{Pr^{.4}} = \frac{\frac{hD_e}{k}}{Pr^{.4}} = \frac{\sigma D_e}{kA(\Delta T)} \times \frac{1}{Pr^{.4}} \qquad (8)$$

where $A = A_u + \gamma A_f$

and $\Delta T = T_w - T_b$

$$Re = \frac{GD_e}{\mu_b} = \frac{4w}{\mu_b P} \tag{9}$$

where $D_e = \frac{4A_c}{P}$

Friction factors for both isothermal and heated flows were computed from Equation (2) using D_e in lieu of D_i.

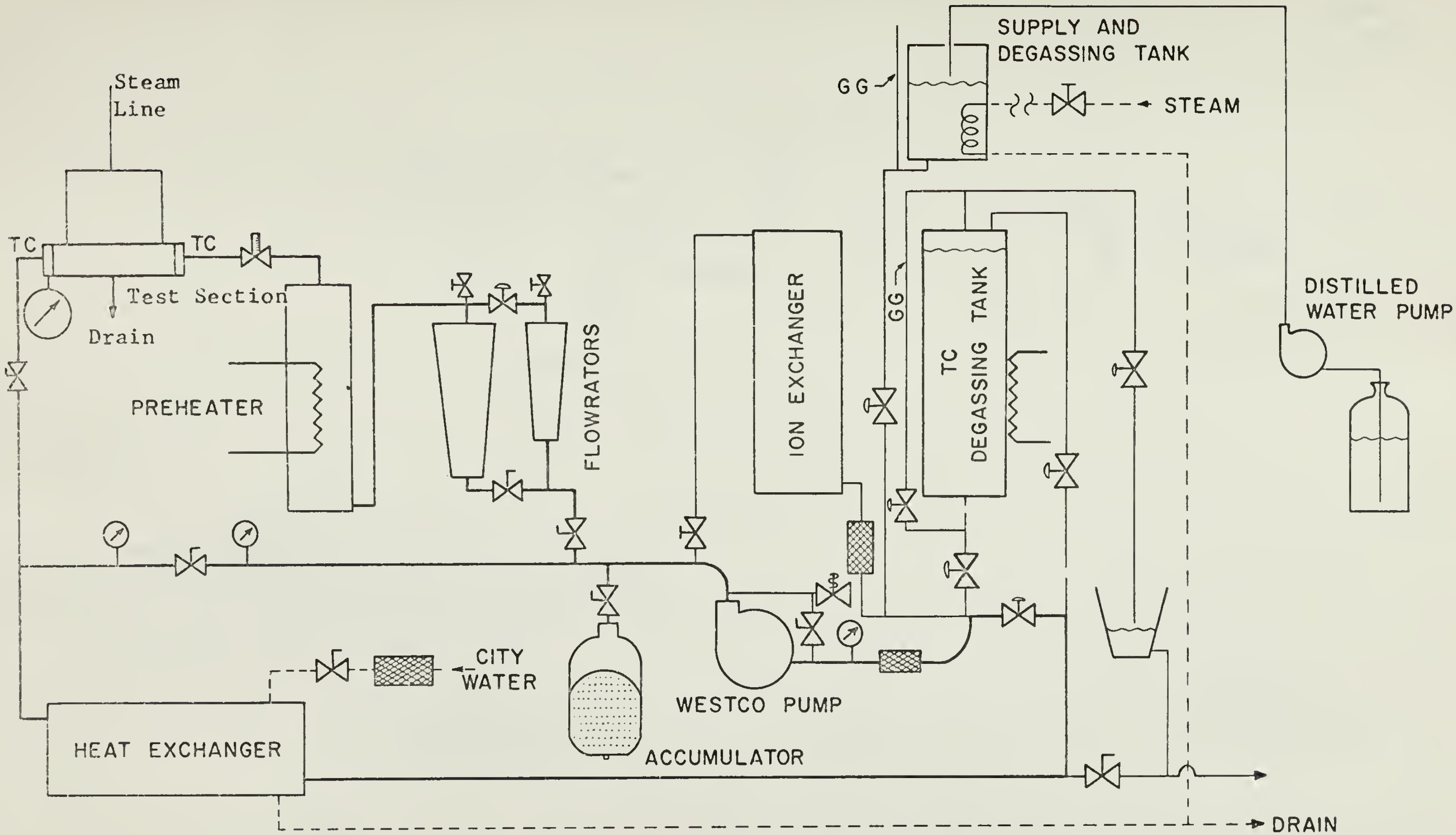

Fig. 1. Schematic of Water Test Loop

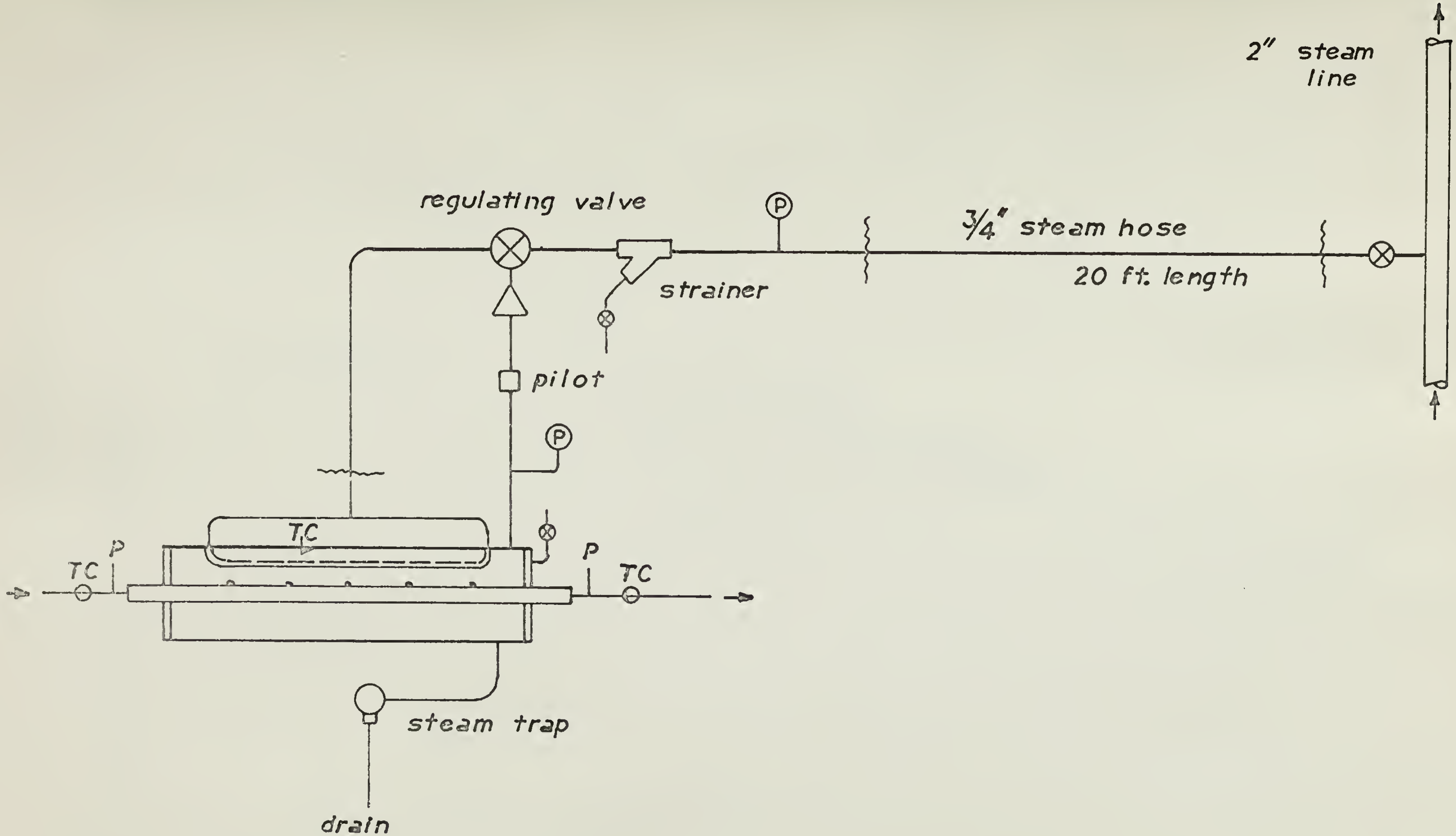

Fig. 2 Schematic of Steam Condensation Apparatus

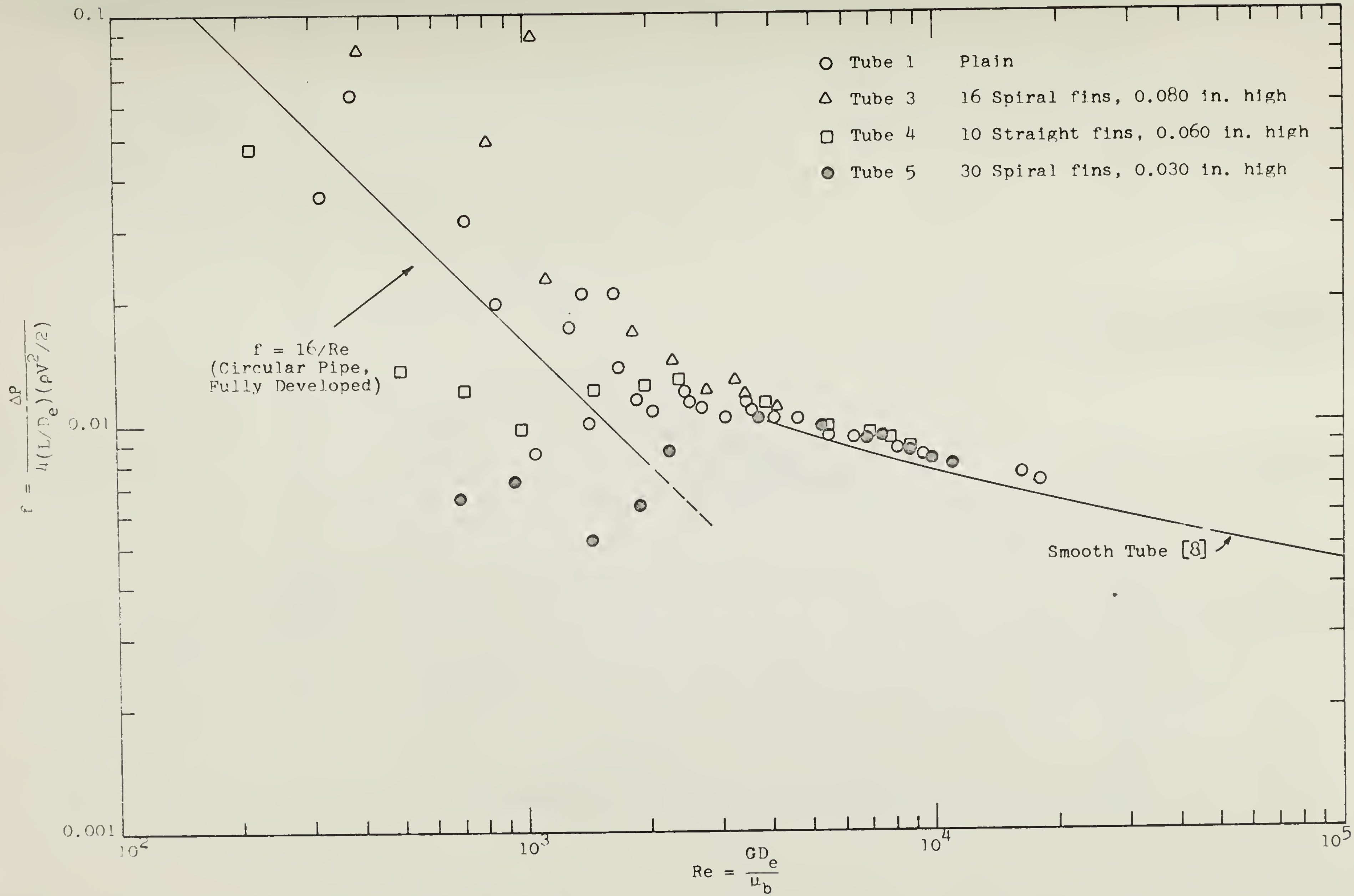

Fig. 3. Friction Factors for Plain and Finned Tubes - Isothermal Conditions

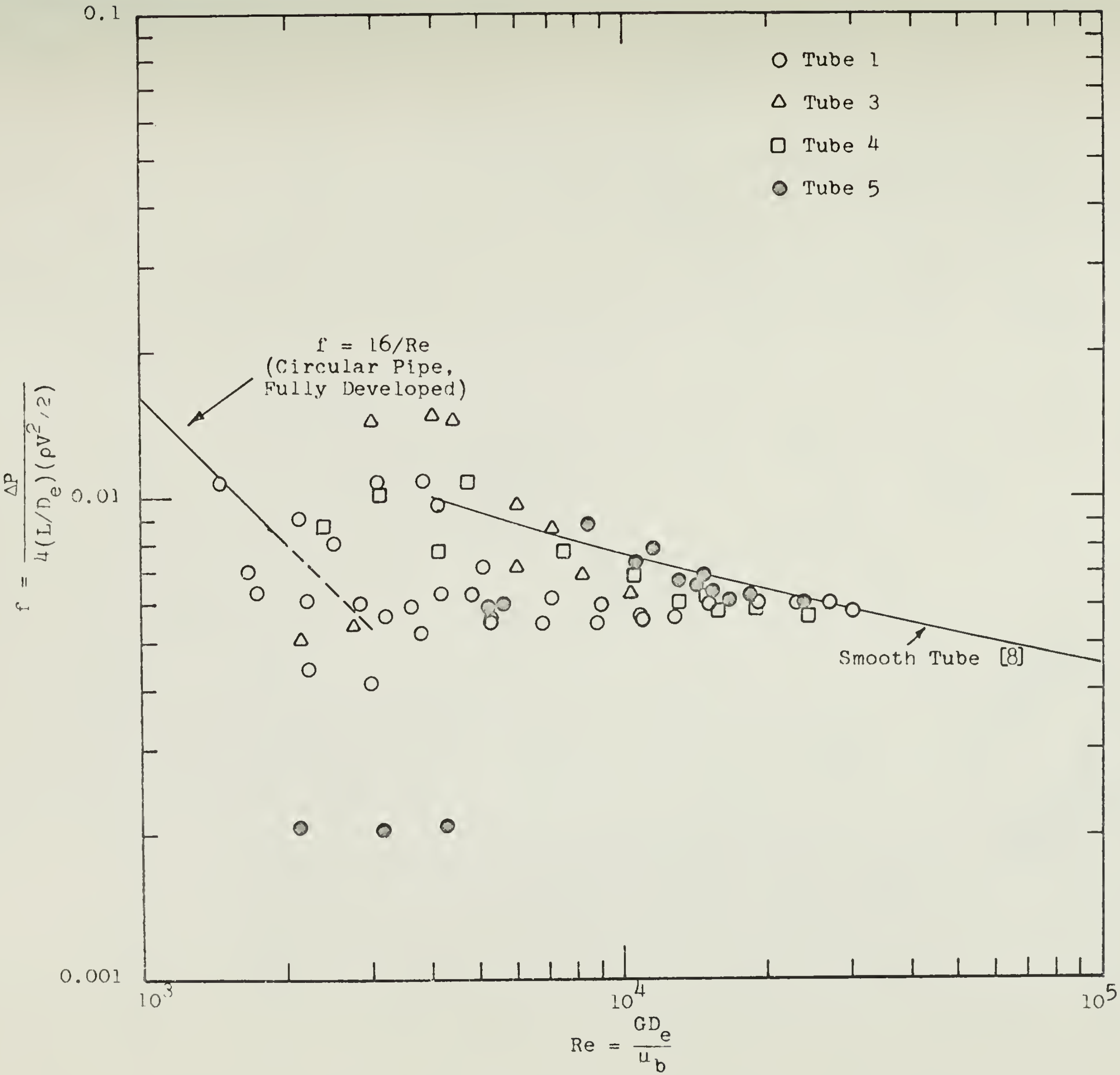

Fig. 4. Friction Factors for Plain and Finned Tubes - Heated Conditions

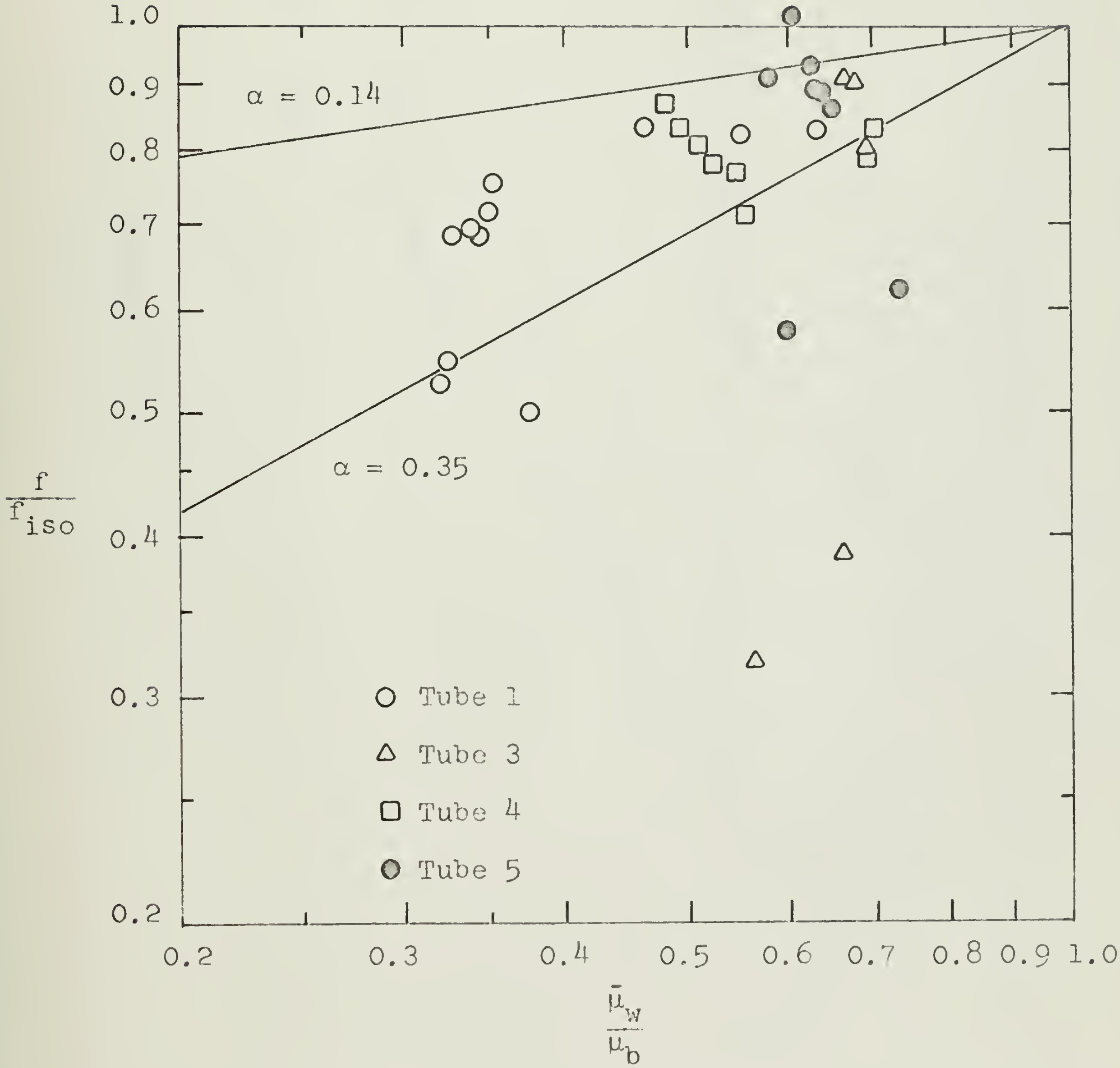

Fig. 5 Correlation of Non-isothermal Friction Factors

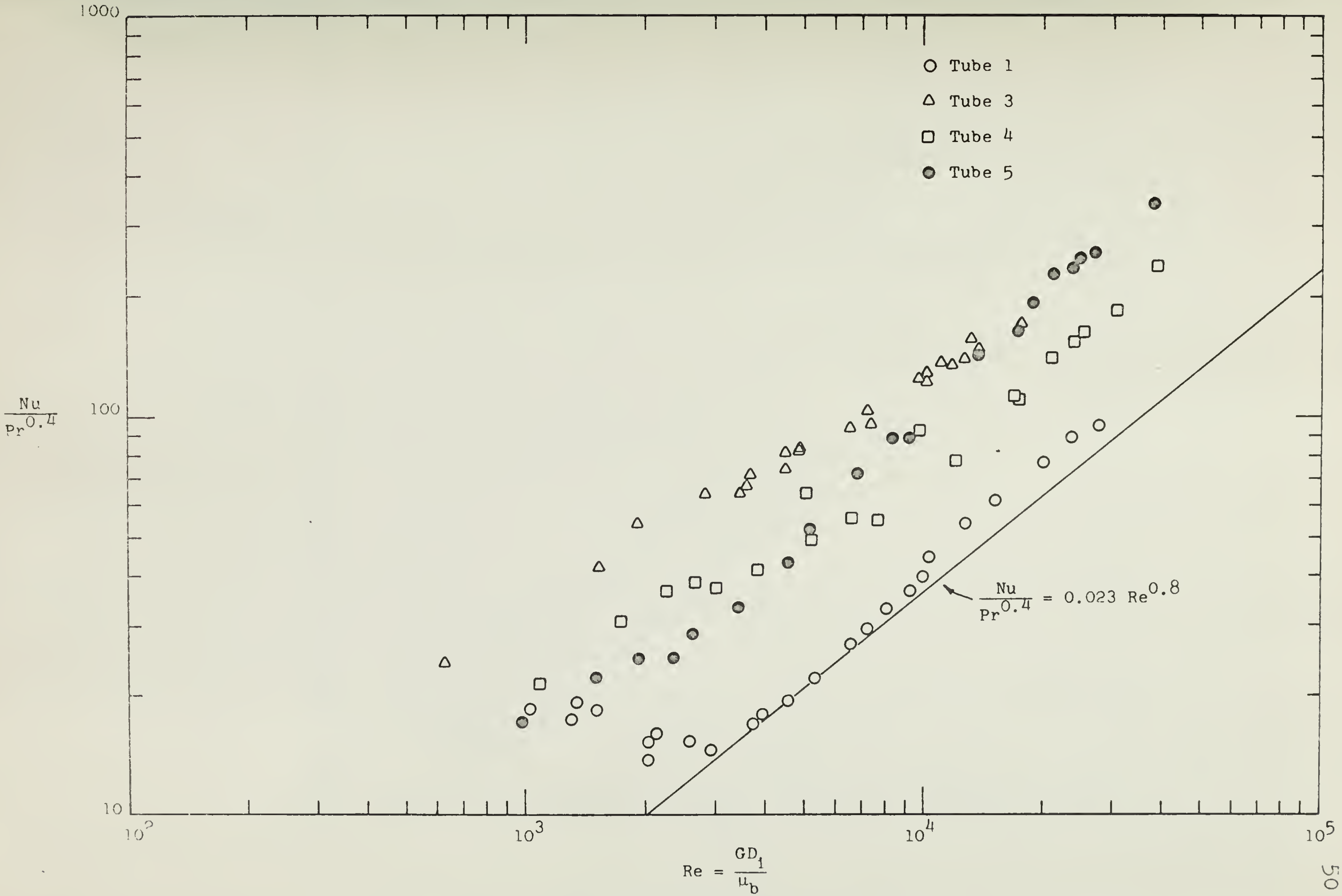

Fig. 6. Nusselt Numbers for Plain and Finned Tubes (Based on Inner Diameter)

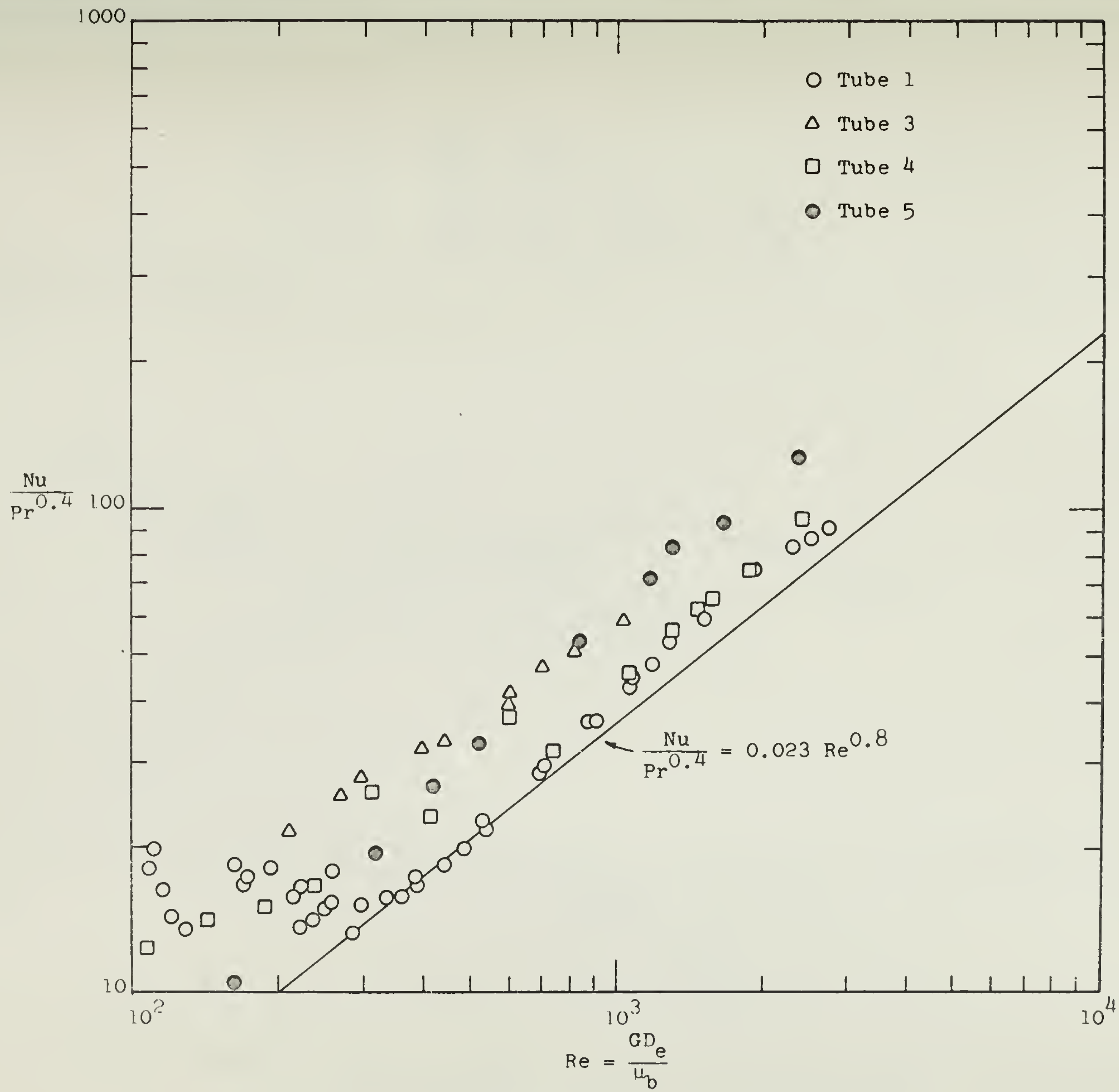

Fig. 7. Nusselt Numbers for Plain and Finned Tubes
(Based on Equivalent Diameter)

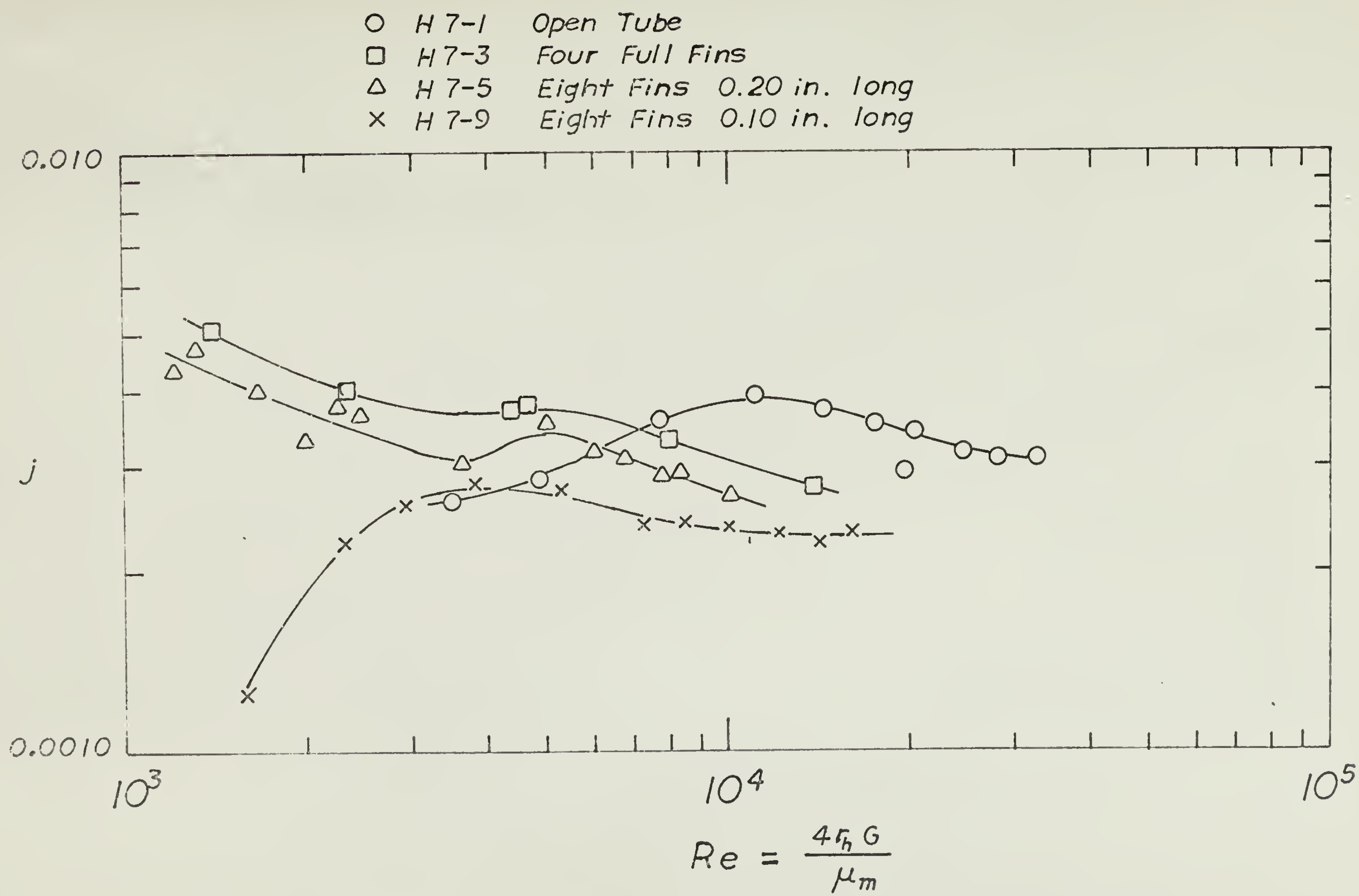

Fig. 8 Dimensionless Correlation of Colburn j-Factor for Internally Finned Tubes. Hilding and Coogan (4)

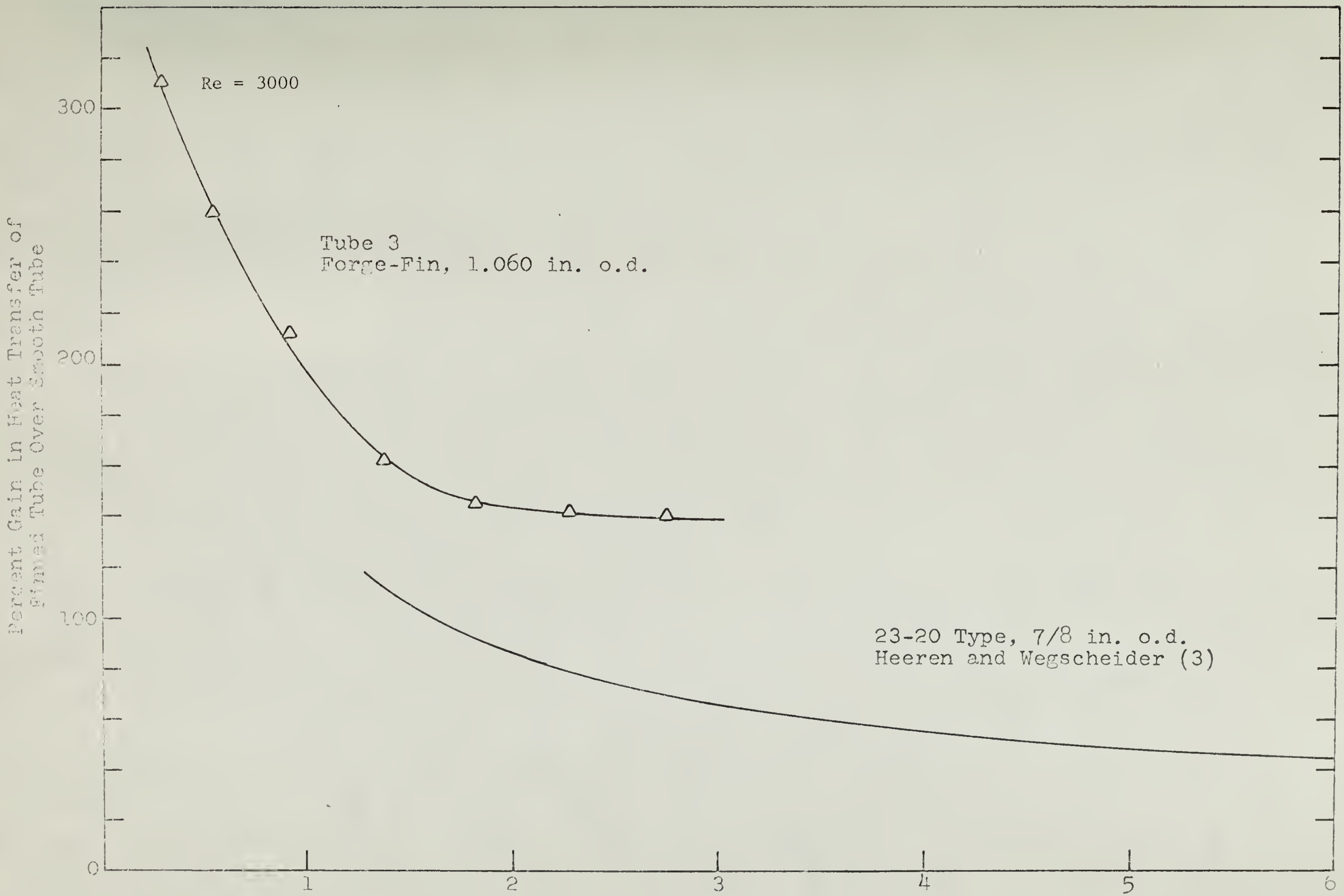

Fig. 9 Gain in Heat Transfer of Finned Tube Over Smooth Tube